JOHN F. KENNEDY SPACE CENTER

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September 15, 1966

LOW TORQUE THREE-PIECE VALVE SEAT

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ABSTRACT

This report describes the tests conducted on a three-piece valve seat designed for high-pressure pneumatic valving.

Three materials were used as seating rings in the assembly. All materials performed satisfactorily from 6000 to 10,000 psig and demonstrated that, for purposes of design, the choice of seating materials is not particularly critical.

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INTRODUCTION

The increased use of pneumatic valving at working pressures ranging from 3,000 to 10,000 psig has led to increased problems with seat-sealing characteristics, seat and seal cold flow and deformation, seating torque, seat erosion under throttled flow conditions, and material stress problems. An analysis of the problems indicate that an ideal high-pressure pneumatic valve seat should have the following characteristics:

a. Resistance to seat erosion under throttled, high-velocity flow conditions of a hard or metal-to-metal seal, without the consequently high seating torque.

b. The leak-tight sealing characteristics of a soft or plastic seat without being susceptible to cold-flow, extrusion, erosion, deformation, and wear failure common to this type of seat.

c. The low-torque seating characteristics of a pressure-assisted soft seal.

A seat designed by KSC Design Engineering (DA) to provide all of these characteristics and the results of tests performed at the high-pressure test facility at Marshall Space Flight Center are described in this report.

APPLICABLE DOCUMENTS

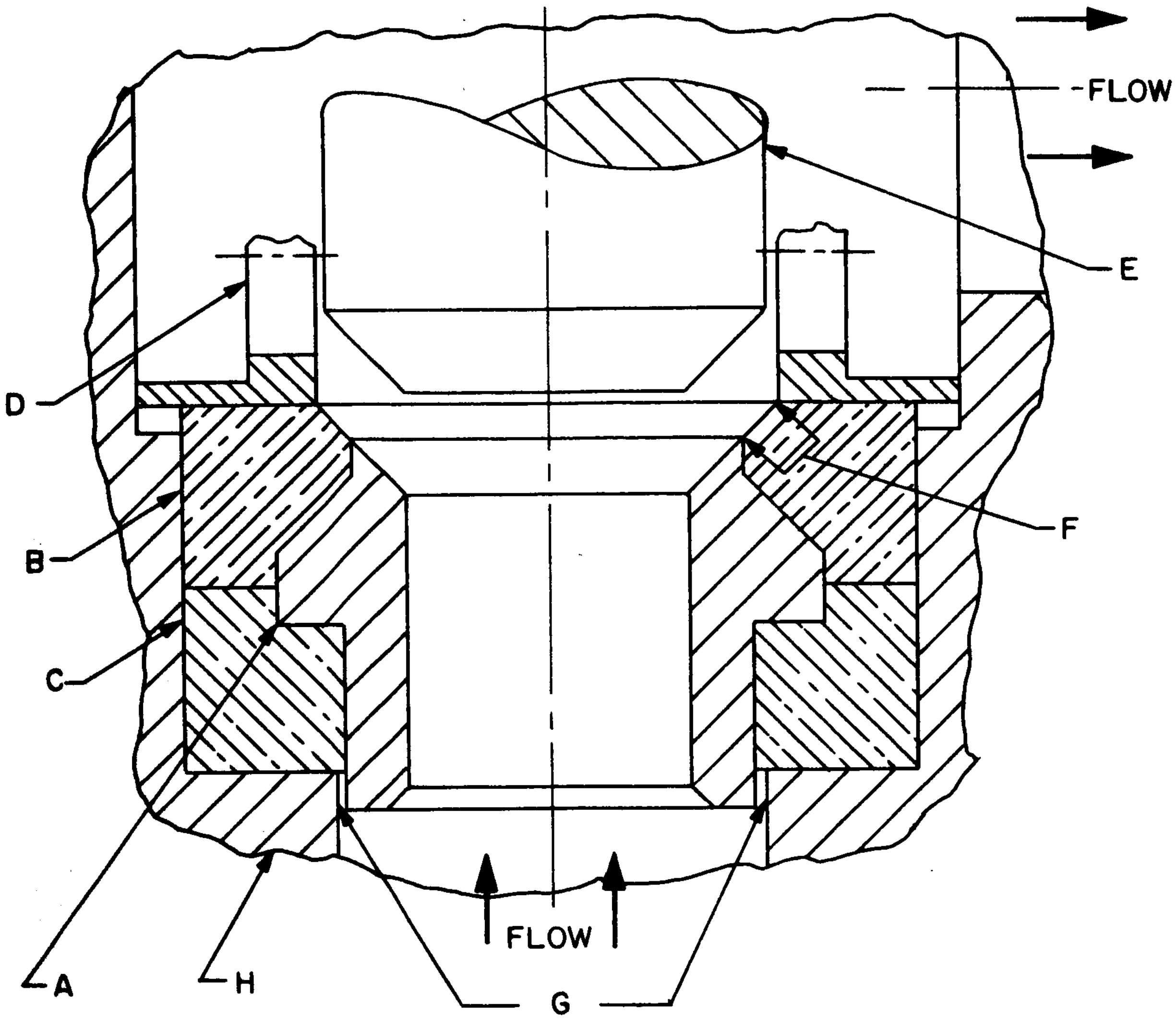
Drawing F-75M883-L
Fabrication Specifications for Three-piece Seat
LOC-23, SK-36

NASA Preliminary Specification 75M09538

DESCRIPTION AND OPERATION

The design geometry of the valve test seat is shown in Figure 1.

Functionally, the seat performs essentially as follows: As plug (E) moves to the closed position and makes contact with upper seat ring (B) and insert ring (A), the unit pressure at the upper seat ring against the plug becomes equal to the flow pressure of the system, due to the system pressure applied to the soft seals through tolerance Gap (G). A further slight closing movement of the plug increases the pressure across sealing area (F) to a value sufficient to ensurea highly leak-tight seal.



A - INSERT RING
B - UPPER SEAT RING
C - LOWER SEAT RING
D - HOLDDOWN RING
E - PLUG
F - LOCAL HIGH-PRESSURE SEALING AREA
G - TOLERANCE GAP
H - VALVE

Figure 1. Cross Section of Typical High-Pressure Valve Seat Configuration

The function of insert ring (A) in the "valve closed" position is to assure that the soft seal rings are as totally confined as possible, to prevent seal cold flow and distortion. During the "near-closed," or throttling position of the plug, the metal insert ring prevents erosion of the soft seal rings by high velocity flow action.

Three test specimens were fabricated to fit a 1/2-inch angle valve which had experienced marked and repeated seat failures during throttling and cycling operation on high-pressure storage batteries at KSC. The design geometry and materials of the three specimens were identical except that upper seat ring (B) was fabricated from Teflon for specimen No. 1, Nylon for specimen No. 2 and Kel-F for specimen No. 3. The lower seat ring (C) was fabricated from Teflon for all three specimens.

QUALIFICATION TESTS

The following tests were performed on the three test specimens. The test setup was as shown in Figure 2. The results of the tests are tabulated in Tables 1 through 7.

Inspection. The body of the test valve was modified to accept the test seats. An inspection of the valve and its specimen seating was made during and after assembly to assure satisfactory fit-up and integrity of the test assembly.

Hydro-Proof Tests. The valve containing test seat specimen No. 1 (Teflon upper ring) was pressurized slowly to a hydrostatic pressure of 10,000 psig with the valve closed to a torque value of 10 inch-pounds and pressure applied in the normal direction of flow. (See Figure 1.) A slight seepage-type leak was noted until the pressure reached 300 psig. At that pressure the leakage stopped and the valve remained leak-tight to 10,000 psig. This pressure was held for 10 minutes and then released. Pressure was then applied in the reverse direction. It was found that an additional 10 inch-pounds of torque was required to seal the valve in this flow direction. A repeat of this reverse pressure test showed that a maximum of 35 inch-pounds of hand wheel torque in the unpressurized condition would assure zero leakage at all pressures from zero to 10,000 psig.

A post-test inspection of the seat assembly revealed that the only effect of the proof tests was a slight flattening of the lower seal ring and small indentations of the upper seat ring in the contact sealing area (F) and in the holddown ring (D) contact areas. These were assessed as normal results to be expected of a "seating-in" process during initial pressurization of a new seat.

Specimens No. 2 and No. 3 were not subjected to a hydrostatic proof test since it was felt that the pneumatic proof tests would be more severe and since the valve test structure had been demonstrated by the hydrostatic tests performed on specimen No. 1.

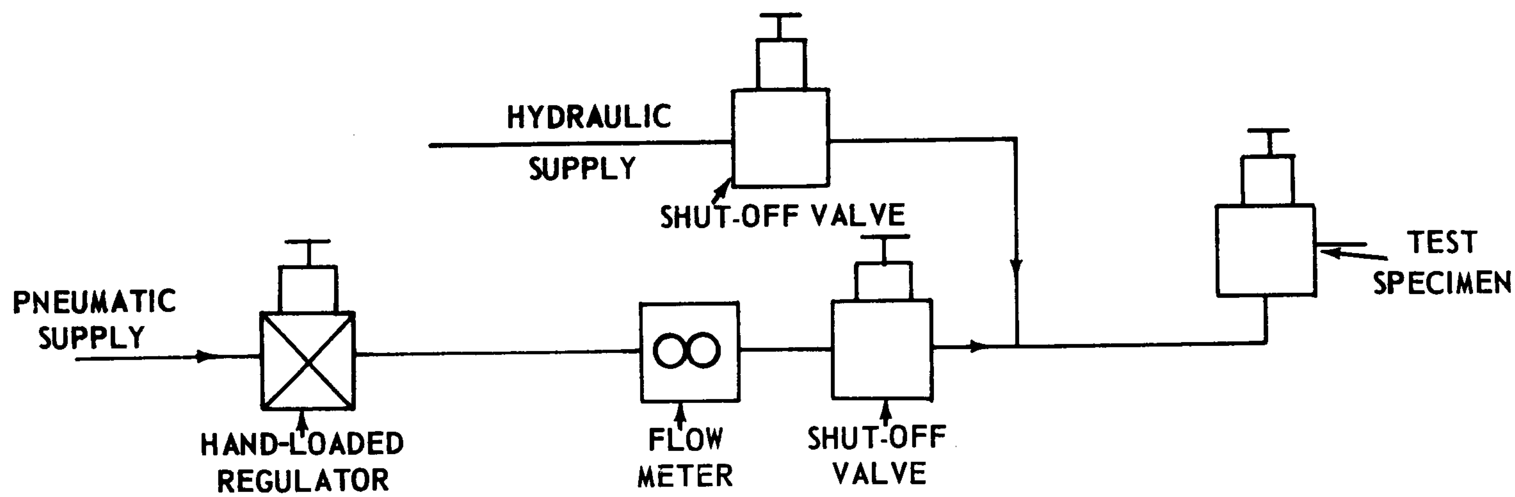


Figure 2. Qualification Tests Setup

Pneumatic Proof Tests. During this test, the valve containing seat specimen No. 1 was pressurized to 10,000 psig with gaseous nitrogen (GN_2) in the normal direction of flow. After this pressure had been attained, the valve was cycled open and then reseated and monitored for leakage for 5 minutes. No leakage was detected and the valve was depressurized.

This cycle of testing was repeated on the valve with specimen No. 2 installed and again with specimen No. 3 installed. All three seat specimens performed equally well at low, handwheel torque values. Post-test inspections revealed no damage to any one of the three seat configurations tested.

Cycle Tests. All three seat configurations were cycle tested at a 6000 psig working pressure using GN_2 as the medium. Specimen No. 1 was cycled 2250 times while specimens No. 2 and No. 3 were cycled 1000 times each. The torque required to seal the valve for specimen No. 1 averaged about 10 inch-pounds; the torque required for specimen No. 2 averaged approximately 25 inch-pounds, and approximately 30 inch-pounds for specimen No. 3. This was felt to be attributable to the differences in the hardness of the upper seal rings. All seat configurations performed equally well during cycling tests. Post-test inspections revealed the seats to be in excellent condition.

Seat Erosion Tests. The three seat specimens were subjected to a seat erosion test which consisted of pressurizing the inlet of the test valve body to 6000 psig with GN_2 and then slightly cracking the valve until a flow of 100 SCFM was attained. This flow rate was maintained for a period of two hours. The specimen was then cycled to the "closed position" to determine any deterioration of the specimen sealing characteristics. All three of the specimens seated leak-tight without any evident increase in handwheel torque over that required for leak-tight seating prior to the erosion test.

A post-test disassembly and inspection revealed no damage or deterioration of any of the three seal configurations.

Table 1. Hydro-Proof Test Results, Specimen No. 1 (Teflon)

INLET PRESSURE	HOLD PERIOD	LEAKAGE	TORQUE	COMMENTS
	Regular Flow direction			
2,000	5 min	Seat	10 inch-pounds	Leak stopped at 300 pounds.
4,000	5 min	None	10 inch-pounds	
6,000	5 min	None	10 inch-pounds	
8,000	5 min	None	10 inch-pounds	
10,000	5 min	None	10 inch-pounds	
	Reverse Flow direction			
2,000	5 min	Seat	10 inch-pounds	Tightened hand valve to stop leakage.
4,000	5 min	None	20 inch-pounds	
6,000	5 min	None	20 inch-pounds	
8,000	5 min	None	20 inch-pounds	
10,000	5 min	None	20 inch-pounds	

Table 2. Pneumatic Proof Test Results, Specimen No. 1 (Teflon)

NO.	PRESSURE*	LEAKAGE	TORQUE	COMMENTS
1	2,000 psi	None	10 inch-pounds	Valve required no more than running torque to close.
2	4,000 psi	None	10 inch-pounds	
3	6,000 psi	None	10 inch-pounds	
4	9,000 psi	None	10 inch-pounds	
5	10,000 psi	None	10 inch-pounds	

* Valve was opened and closed at each pressure increment. Gaseous Helium (GHe) used as pressurant.

Table 3. Pneumatic Proof Test Results, Specimen No. 2 (Nylon)

NO.	PRESSURE*	LEAKAGE	TORQUE	COMMENTS
1	2,000 psi	None	26 inch-pounds	
2	4,000 psi	None	27 inch-pounds	
3	6,000 psi	None	28 inch-pounds	
4	8,000 psi	None	30 inch-pounds	
5	10,000 psi	None	30 inch-pounds	

* Gaseous Helium (GHe) used as pressurant.

Table 4. Pneumatic Proof Test Results, Specimen No. 3 (Kel -F)

NO.	PRESSURE*	LEAKAGE	TORQUE	COMMENTS
1	2,000 psi	None	20 inch-pounds	
2	4,000 psi	None	20 inch-pounds	
3	6,000 psi	None	24 inch-pounds	
4	8,000 psi	None	26 inch-pounds	
5	10,000 psi	None	25 inch-pounds	

* Gaseous Helium (GHe) used as pressurant.

Table 5. First Cycle Test Results (Using Teflon Upper Seat)

NO.	CYCLES	PRESSURE	TORQUE	LEAKAGE	COMMENTS
1	50	6,000 psi	10 inch-pounds	None	Torque did not increase as the cycles progressed.
2	100	6,000 psi	10 inch-pounds	None	
3	150	6,000 psi	10 inch-pounds	None	
4	200	6,000 psi	10 inch-pounds	None	
5	250	6,000 psi	50 inch-pounds	325 SCIM	
	NOTE: Additional torque did not slow down leakage across seat. Inspection showed the balancing O-ring to be worn and leaking.				Changed O-rings
6	200	6,000 psi	10 inch-pounds	None	
7	400	6,000 psi	10 inch-pounds	None	
8	600	6,000 psi	10 inch-pounds	None	
9	800	6,000 psi	10 inch-pounds	325 SCIM	Changed O-rings
10	1,000	6,000 psi	10 inch-pounds	None	
11	1,200	6,000 psi	15 inch-pounds	None	
12	1,400	6,000 psi	15 inch-pounds	325 SCIM	Changed O-rings
13	1,600	6,000 psi	15 inch-pounds	None	
14	1,800	6,000 psi	15 inch-pounds	None	
15	2,000	6,000 psi	15 inch-pounds	None	

Table 6. Second Cycle Test Results

NO.	CYCLES	PRESSURE	TORQUE	LEAKAGE	COMMENTS
Using Kel-F Upper Seat					
1	100	6,000 psi	20 inch-pounds	None	
2	200	6,000 psi	20 inch-pounds	None	
3	300	6,000 psi	20 inch-pounds	None	
4	400	6,000 psi	20 inch-pounds	None	
5	500	6,000 psi	25 inch-pounds	None	Torque to Close increased.
6	600	6,000 psi	25 inch-pounds	None	
7	700	6,000 psi	25 inch-pounds	None	
8	800	6,000 psi	25 inch-pounds	None	
9	900	6,000 psi	25 inch-pounds	None	
10	1,000	6,000 psi	25 inch-pounds	None	
Using Nylon Upper Seat					
1	100	6,000 psi	30 inch-pounds	None	Valve had audible leak until 50 inch-pounds was applied; once seal was made, 30 inch-pounds resealed it.
2	200	6,000 psi	30 inch-pounds	None	
3	300	6,000 psi	30 inch-pounds	None	
4	400	6,000 psi	30 inch-pounds	None	
5	500	6,000 psi	30 inch-pounds	None	
6	600	6,000 psi	30 inch-pounds	None	
7	700	6,000 psi	30 inch-pounds	None	
8	800	6,000 psi	30 inch-pounds	None	
9	900	6,000 psi	30 inch-pounds	None	
10	1,000	6,000 psi	30 inch-pounds	None	

Table 7. Seat Erosion Test Results

PRESSURE	FLOW RATE	PERIOD	COMMENTS
		Using Teflon Seat	
6,000 psi	100 SCFM	2 hours	Torqued closed at 12 inch-pounds.
		Using Kel-F Seat	
6,000 psi	100 SCFM	4 hours	Torqued closed at 25 inch-pounds.
		Using Nylon Seat	
6,000 psi	100 SCFM	2 hours	Torqued closed at 30 inch-pounds.

Inspection: No damage to parts. No notable marks resulted from the poppet meeting the seat.

CONCLUSIONS

Three-piece valve seat configurations using Teflon, Kel-F, and Nylon as the seat ring material successfully demonstrated a capability to seat under cyclic and seat erosive conditions at 6000 psig pneumatic working pressures and to withstand pneumatic proof pressures "across the seat" of 10,000 psig.

The test results indicate that a variety of materials can be used successfully for seating materials, such as soft metals and various grades of plastics, thus permitting the selection of seating material to be based on design factors, such as system compatability, temperature, strength, etc.

Several O-ring failures occurred in the valve test body mechanisms during the testing, and while these failures are not related to the test proper of the valve seat, it should be pointed out that the use of O-rings as dynamic seals in an extremely dry gaseous system should be approached with caution.

The capability parameters of the three-piece valve seat were shown to be as follows:

a. Pneumatic pressure holding capability "across the seat" of 10,000 psig.

b. A capability of a minimum 1000 cycles of operation (open and close) at a pneumatic working pressure of 6000 psig.

c. Seat erosion resistance to a throttled pneumatic flow of 100 SCFM with an upstream pressure of 6000 psig.

The torque values required to achieve seating were extremely low and the pressure-assisted sealing capabilities were within design expectations.

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VALVE SEAT

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